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ABSTRACT

A probabilistic two-phase flow map condensation heat transfer model for R134a in 8mm ID tube is developed in order to more easily predict heat transfer in multiple flow regimes with the same flow regime basis that can be used to model pressure drop and void fraction. Continuous time fraction curve fits of the intermittent, stratified and annular flow regimes with physically correct limits were made for probabilistic two-phase flow maps found in the literature. The time fraction functions are simply multiplied by a respective model developed for each flow regime and summed toobtained the predicted heat transfer coefficient. The developed model is compared to experimentally obtained condensation data and models found in the literature. It is determined that the definition of the annular flow regime or the annular flow regime models used must be modified in order to eliminate inaccuracies of the present model at low quality ranges.

1. INTRODUCTION

Numerous two-phase flow heat transfer models can be found in the literature. Many of the models can be categorized as stratified flow models such as Chato's (1962) classic model, or annular flow models. The annular flow heat transfer models can be further divided into two-phase multiplier based models such as Shah (1979), shear based models such as Soliman (1968), and boundary layer correlation based models such as Traviss et al. (1973) and Hurlburt and Newell (1999). These heat transfer models are based on assumptions of the nature of the two phase flow as a whole.

Recently, flow regime map based two-phase flow heat transfer models have been developed by Wojtan et al. (2005b)), Thome et al. (2003), Cavallini et al. (2002), Zurcher et al. (2002). These two-phase flow regime map based heat transfer models are found to predict heat transfer for multiple flow regimes and for a wide range of fluid properties. All of these models utilize Steiner type flow regime maps which indicate a particular flow regime at a given quality and mass flux. The distinct lines seem to lack a physical basis as Coleman and Garimella (2003), El Hajal et al. (2003), and Niño (2002) indicate that more than one flow regime can exist near the boundaries or within a given flow regime on a Steiner type flow map. Steiner type flow maps can not be represented as continuous functions for the entire quality range as can be seen from a flow map recently developed by Wojtan et al. (2005a) and presented in figure 1. Consequently, these heat transfer models tend to be complicated and they require an interpolation to avoid discontinuities at the flow regime boundaries.

Probabilistic two-phase flow regime maps first developed by Niño (2002) for refrigerant and air-water flow in multi-port microchannels are found by Jassim and Newell (2006) to eliminate the discontinuities created by traditional flow maps. Probabilistic two phase flow regime maps have quality on the horizontal axis and the fraction of time in which a particular flow regime is observed in a series of pictures taken at given flow condition (F) on the y axis as seen in figure 2. Jassim and Newell (2006) developed curve fit functions to represent the data that are continuous for the entire quality range with correct physical limits for the time fraction data obtained for 6-port microchannels by Niño (2002). Jassim and Newell (2006) then utilized the probabilistic flow regime map time fraction curve fits to predict pressure drop and void fraction as shown in equations 1 and 2 respectively.

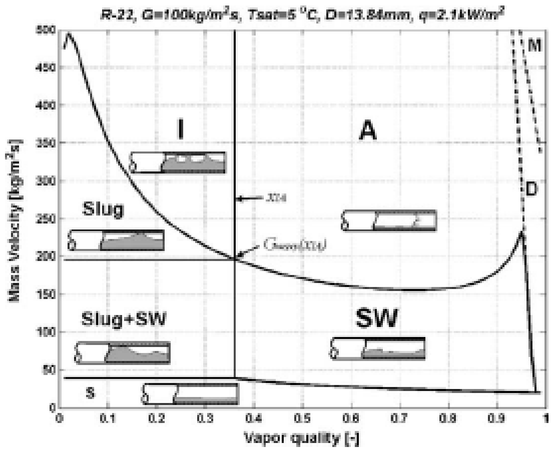

Figure 1. Typical Steiner type flow map taken from Wojtan et al. (2005)

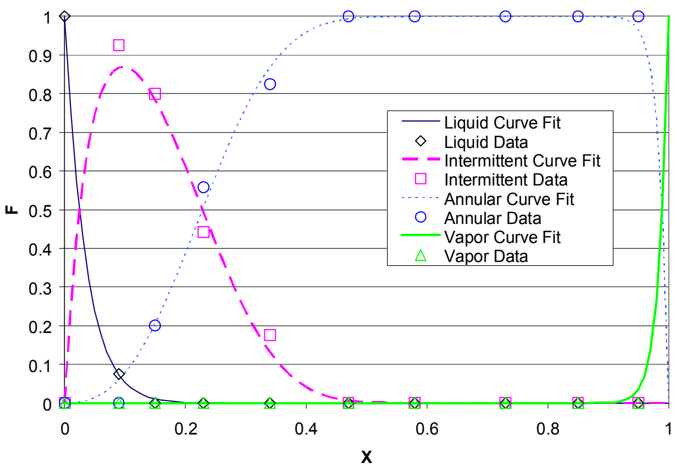

Figure 2. Probabilistic flow map with time fraction curve fits for R410A, 10^o C, 300kg/m^2-s in a 6-port 1.54 mm hydraulic dia. microchannel taken from Jassim and Newell (2006).

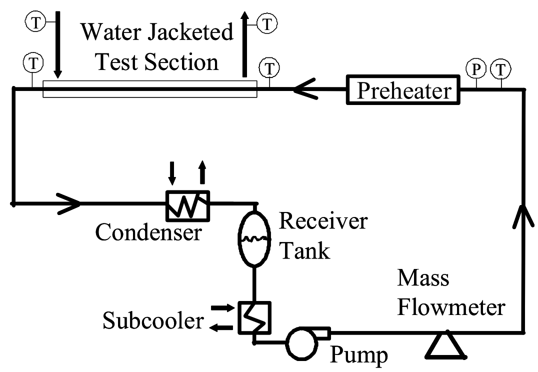

Figure 3. Two-phase flow loop schematic

$$\left(\frac{dP}{dz}\right)_{total} = F_{liq}\left(\frac{dP}{dz}\right)_{liq} + F_{\text{int}}\left(\frac{dP}{dz}\right)_{\text{int}} + F_{vap}\left(\frac{dP}{dz}\right)_{vap} + F_{ann}\left(\frac{dP}{dz}\right)_{ann} \tag{1}$$

$$\alpha_{total} = F_{liq}\alpha_{liq} + F_{\text{int}}\alpha_{\text{int}} + F_{vap}\alpha_{vap} + F_{ann}\alpha_{ann} \tag{2}$$

In this way pressure drop and void fraction models developed for a particular flow regime are easily and properly weighted for the entire quality range on a consistent basis. Jassim et al. (2006) recently developed probabilistic two-phase flow regime maps for 1.7 to 8mm ID smooth adiabatic tubes with both R134a and R410A, and a wide range of mass fluxes and qualities.

In the present study probabilistic two-phase flow maps developed by Jassim et al. (2006) are utilized for the prediction of condensation heat transfer in smooth horizontal tubes. Curve fit functions are developed to represent the time fraction data for the intermittent/liquid, stratified, and annular flow regimes with physically correct limits. A probabilistic two phase heat transfer model is developed which utilizes the time fraction curve fits in a manner consistent with the pressure drop and void fraction models give in equations 1 and 2 respectively. This model is compared to experimentally obtained heat transfer data of R134a at 25°C in 8.9mm ID smooth copper tube under condensation conditions at 100, 200, and 300 kg/m^2-s. The developed model is also compared to flow regime map based condensation models found in the literature.

2. EXPERIMENTAL TWO-PHASE HEAT TRANSFER TEST FACILITIES AND TEST MATRIX

The heat transfer data was obtained from the two-phase flow loop depicted in figure 3. The liquid refrigerant is pumped with a gear pump that is driven by a variable frequency drive from the bottom of a 2 liter receiver tank through a water cooled shell and tube style subcooler in order to avoid pump cavitation. The liquid refrigerant then travels through a corriolis style mass flow meter followed by a preheater used to reach the desired quality. The preheater consists of a finned tube heat exchanger with opposing electric resistance heater plates bolted on either side of the heat exchanger. The electric heaters are controlled with on/off switches and a variac to provide fine adjustment of quality. This preheater design has sufficient thermal mass to avoid burn out at a quality of 100% and has a sufficiently small thermal mass so that steady state conditions can be rapidly attained. The refrigerant is then directed through 90 degree bends to remove effects of heat flux from the preheater such as dryout before it reaches the test section.

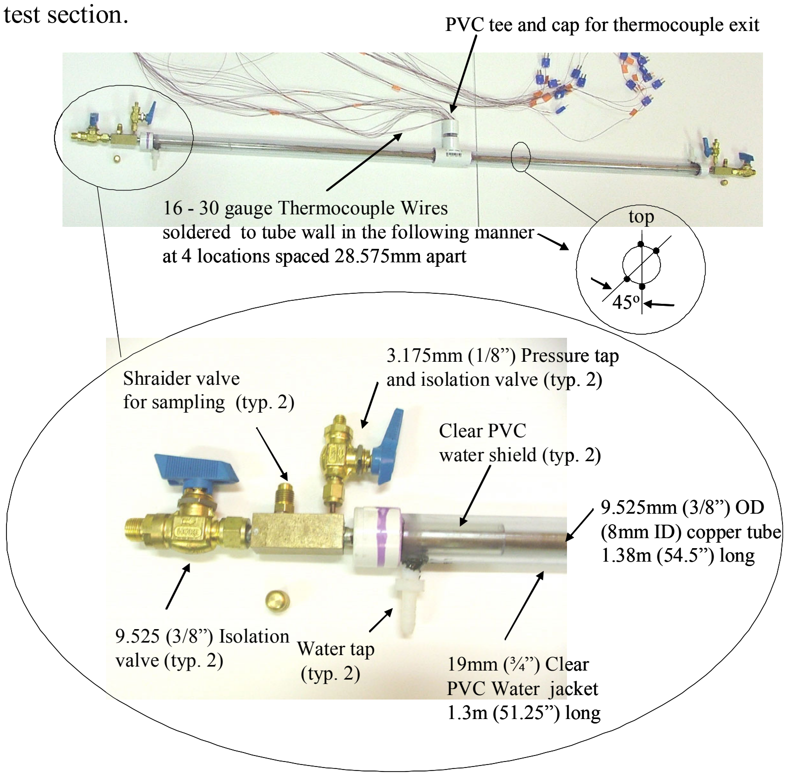

Figure 4. Test section design

The test sections consists of an 8.9mm I.D. copper tube with 4 stations of 4 thermocouples welded on the walls and a clear P.V.C water jacket as seen in figure 4. The wall temperature is determined as the average of the 16 thermocouples welded on the tube wall. Insulation is placed on the outside of the water jacket. The test section is cooled with water supplied from a water conditioner at 15.7^{o}C. Thermocouples are placed at the inlet and outlet of the test section in order to determine the average refrigerant temperature. A flow meter is placed in the water jacket loop and thermocouples are placed at the inlet and exit of the test section water jacket in order to determine the heat lost by the test section. After the test section the refrigerant is condensed in a flat plate heat exchanger with cold water at 5^o C. The loop temperature is controlled by varying the flow rate of the cold water entering the condenser. Condensation heat transfer measurements were obtained for R134a at 25^{o}C saturation temperature, mass fluxes of 100, 200, and 300kg/m^2-s and a heat flux ranging from 4,000 to 6,000 W/m^2.

3. HEAT TRANSFER MODEL DEVELOPMENT

3.1 Curve fitting of probabilistic two-phase flow regime maps

Probabilistic two-phase flow regime maps obtained from Jassim et al. (2006) are used to develop time fraction curve fits that can be utilized to create a new heat transfer model. Time fraction functions are developed that are continuous from a quality of 0 to 1 and represent the intermittent/liquid, stratified, and annular flow regimes. The intermittent/liquid flow regime time fraction function, given in equation 3, is continuous from a quality of 0 to 1 and has the correct physical limits with a time fraction of one at a quality of zero and a time fraction of zero at a quality of one.

$$F_{\mathrm{int}+liq} = (1-x)^a \tag{3}$$

The stratified flow regime time fraction function, given in equation 4, is also a continuous function with correct physical limits with a time fraction of zero at a quality of zero and one.

$$F_{strat} = \left(1 - x^{(b-x)^c}\right)^a - (1-x)^a \tag{4}$$

The annular flow regime time fraction function, given in equation 5, is simply one minus the time fraction of the other flow regimes.

$$F_{ann} = 1 - F_{\mathrm{int}+liq} - F_{strat} \tag{5}$$

These time fraction functions reasonably represent the data as seen in figures 5, 6, and 7 for R134a in 8mm ID smooth adiabatic tube at 25^o C and 100, 200, and 300kg/m^2-s respectively. The curve fit constants used for each mass flux are summarized in table 1.

Table 1. Curve fit constants for time fraction functions of R134a at 25^{o}c in 8mm ID smooth adiabatic tube

Mass flux in kg/m^2-s	a	b	c
100	18.6	10.0	20.0
200	24.0	1.2	5.4
300	28.7	1.1	5.9

3.2 Probabilistic two-phase flow map heat transfer model

A physically based heat transfer model can be created, given in equation 6, with the probabilistic two-phase flow regime map information contained in the continuous time fraction functions.

$$h_{total} = F_{\mathrm{int}+liq} h_{\mathrm{int}+liq} + F_{strat} h_{strat} + F_{ann} h_{ann} \tag{6}$$

The intermittent flow regime heat transfer model $h_{\mathrm{int+liq}}$, defined in equation 7, is a modified Dittus Boelter relation which uses the liquid properties and the total two-phase mass flux.

$$h_{\mathrm{int}+liq} = 0.023\left(\frac{k_l}{D}\right)\left(\frac{GD}{\mu_l}\right)^{0.8} \mathrm{Pr}_l^{\,0.3}, \text{ where } \mathrm{Pr}_l = \frac{\mu_l Cp_l}{k_l} \tag{7}$$

The stratified flow regime heat transfer model h_{strat}, defined in equation 8, is simply the Chato (1962) model for stratified flow.

$$h_{strat} = 0.555\left(\frac{k_l}{D}\right)\left(\frac{\rho_l(\rho_l-\rho_v)gh_{lv}D^3}{k_l\mu_l(T_{sat}-T_{wall})}\right)^{0.25} \tag{8}$$

The annular flow regime heat transfer model h_{ann}, defined in equation 9, is the annular flow regime model presented in Dobson and Chato (1998).

$$h_{ann} = 0.023\left(\frac{k_l}{D}\right)\mathrm{Re}_l^{0.8}\,\mathrm{Pr}_l^{0.4}\left(1+\frac{2.22}{X_{tt}^{\;0.889}}\right), \text{ where } X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1} \tag{9}$$

The heat transfer model presented in equation 6 is flexible because it is able to accept different heat transfer models for each flow regime. The heat transfer models used for each flow regime presented in equations 7 through 9 are chosen for their simplicity.

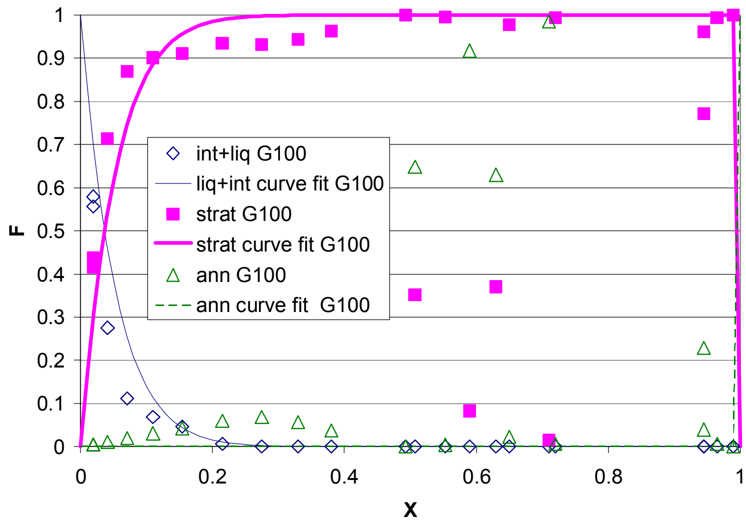

Figure 5. Probabilistic two-phase flow map for R134a, 25°C, 100kg/m^2-s, 8mm ID smooth adiabatic tube

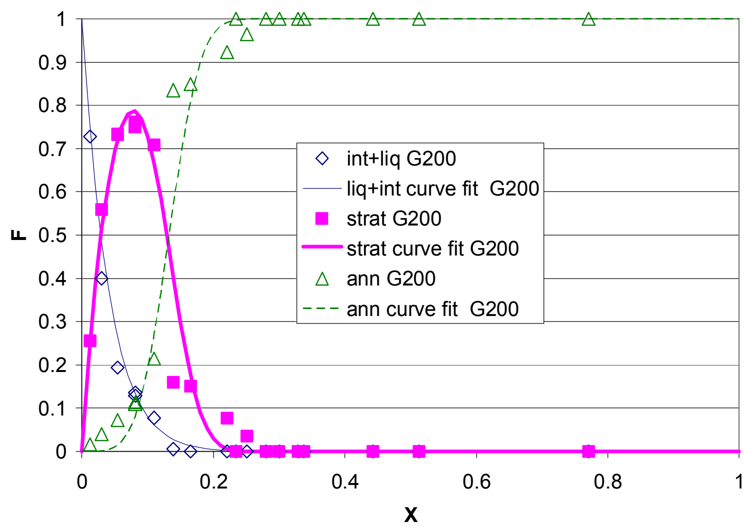

Figure 6. Probabilistic two-phase flow map for R134a, 25°C, 200kg/m^2-s, 8mm ID smooth adiabatic tube

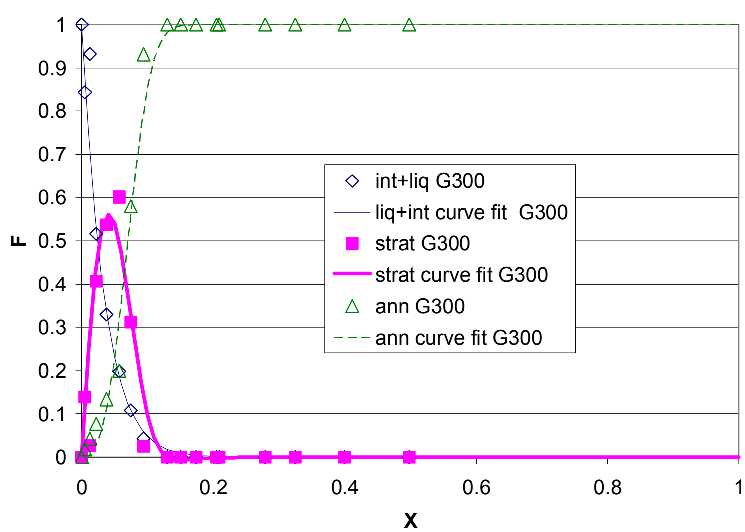

Figure 7. Probabilistic two-phase flow map for R134a, 25°C, 300kg/m^2-s, 8mm ID smooth adiabatic tube

4. RESULTS

The experimentally obtained condensation heat transfer coefficients for R134a at 25° C in smooth tubes for 100, 200, and 300kg/m^2-s are plotted for a range of qualities in figures 8, 9, and 10 respectively. The transfer models of Thome et al. (2003), Cavallini et al. (2002), the probabilistic model presented in equation 6, and the probabilistic model presented in equation 6 with the Thome et al. (2003) model for annular flow replacing equation 9. Since the time fraction curve fits for the 5.4 and 8mm ID tubes are similar it is reasonable to compare the probabilistic model for the 8mm ID tube with the heat transfer data for the 8.9mm ID smooth tube.

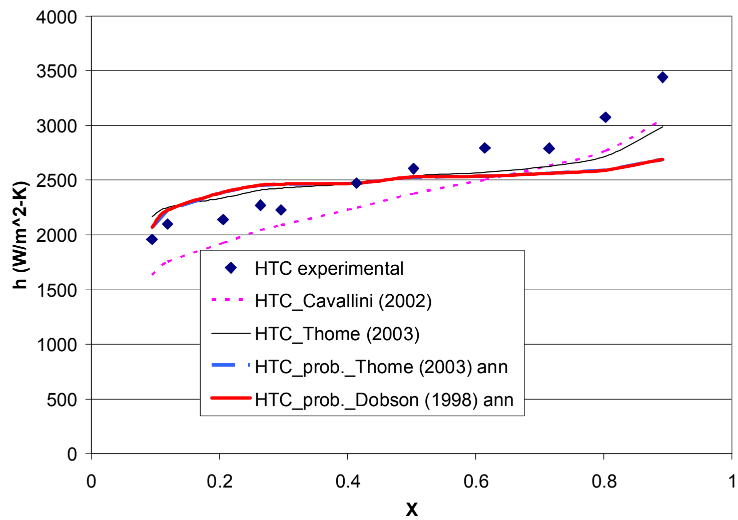

Figure 8. h for R134a, 25°C, 100kg/m^2-s, in 8.9 mm ID smooth tube, condensation varied 4400-5200 W/m^2

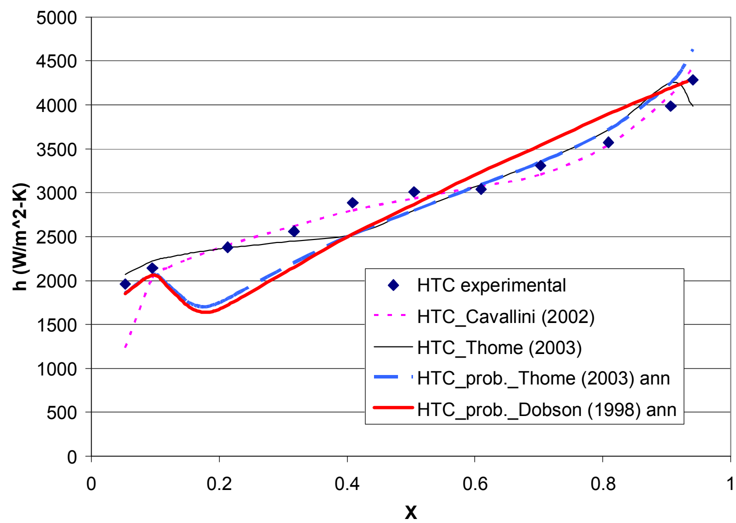

Figure 9. h for R134a, 25°C, 200kg/m^2-s, in 8.9 mm ID smooth tube, condensation varied 4500-5400 W/m^2

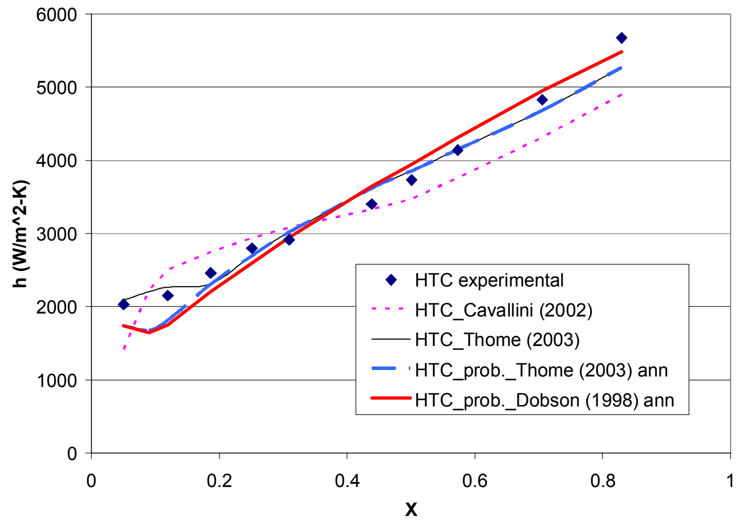

Figure 10. h for R134a, 25°C, 300kg/m^2-s, in 8.9 mm ID smooth tube, condensation varied 4400-5300 W/m^2

The experimental data in figures 8, 9, and 10 is seen to lie close to the models developed by Thome et al. (2003) and Cavallini et al. (2002). The probabilistic models developed, one incorporating Dobson and Chato's (1998) annular flow model, and the other incorporating the annular flow model of Thome et al. (2003), represent the data well at high and very low quality ranges for the 200 and 300 kg/m^2-s curves. The uncertainty in the heat transfer data is found to be a maximum of $\pm$11.5%, the uncertainty in quality is less than $\pm$ 0.02 at x=0.99, $\pm$ 0.012 at x=.5, and +0.002 at x=0.02, and the uncertainty in mass flux is less than $\pm$2%. However, there is a dip in heat transfer at a quality of 0.2 and 0.1 in figures 9 and 10 respectively. This observation may be the result of the definition of annular flow used in Jassim et al. (2006) as a state where refrigerant is present at the top of the tube wall. An alternate definition of annular flow or a different annular flow model should be used in the future to eliminate the dip in heat transfer coefficients.

5. CONCLUSION

In summary, probabilistic two-phase flow map heat transfer models are developed in a similar manner as pressure drop and void fraction models have been developed in the literature for multi-port microchannels. Previously developed probabilistic two-phase flow regime maps are utilized to develop continuous time fraction curve fits for the intermittent, liquid, and annular flow regimes with physically correct limits. The probabilistic two-phase flow map heat transfer model weights the importance of each flow regime based on the time fraction functions. The present model is compared with experimentally obtained heat transfer data and flow map based models found in the literature with reasonable agreement. However, the definition of the annular flow regime or the annular flow regime model used must be modified in order to eliminate inaccuracies of the present model at low quality ranges.

NOMENCLATURE

Cp	specific heat (kJ/kg-K)	h_{lv}	heat of vaporization (kJ/kg)
D	tube inner diameter (m)	k	thermal conductivity (W/m)
dP	pressure drop (Pa)	Re	Reynolds number (-)
dz	unit length (m)	T	temperature (K)
F	observed time fraction (-)	x	flow quality (-)
G	mass flux (kg/m^2-k)	X_{tt}	Lockhart-Martinelli parameter (-)
h	heat transfer coefficient (W/m^2-K)		

Greek symbols

α	void fraction (-)	Pr	Prandlt Number
μ	dynamic viscosity (kg/m-s)	ρ	density (kg/m^3)

Subscripts

ann	pertaining to the annular flow regime	*sat*	saturation
int	pertaining to the intermittent flow regime	*v*	vapor
liq	pertaining to the liquid flow regime	*vap*	pertaining to the vapor flow regime
l	liquid		

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